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A STUDY OF PISTON AND RING FRICTION

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SUMMARY

The apparatus used in this investigation permits the isolation of piston and ring friction from total engine friction under actual operating conditions. Friction work has been measured for various piston ring combinations and cylinder surfaces. The effect of scuffing and excess cylinder-wall lubrication on piston and ring friction is reported, and a careful analysis of the changes in friction work during the run-in period is presented. A new technique for obtaining photomicrographs of curved surfaces has been used to make visual comparisons of cylinder-wall roughness. A device for measuring diametral ring tensions and an apparatus for transforming pressure-crank angle indicator diagrams into pressure-volume indicator diagrams are described.

INTRODUCTION

The work described in this report is a continuation of the work of Forbes and Taylor (reference 1), who built an apparatus and devised a method for isolating piston and ring friction from other forms of engine friction.

The work reported in reference 1 consisted mainly of the development of a reasonably satisfactory apparatus, but produced only preliminary results in the way of friction data. The material presented herewith describes further improvement of the apparatus and the results of test runs on a number of different piston and ring combinations.

In addition to the work described herein and in reference 1, the work of Tischbein (reference 2) in Germany, and Hawkes and

Hardy (reference 3), in England, may be mentioned. The apparatus used by Tischbein was similar to that to be described in this report, insofar as the frictional drag of the piston rings was measured by the deflection of a flexibly mounted cylinder, but differed by having a cross head which allowed the piston to move without touching the cylinder walls and thereby permitted the measurement of ring friction only. The apparatus of Hawkes and Hardy was also in the same category, except that the piston was flexibly mounted and the cylinder sleeve was caused to oscillate. Although Hawkes and Hardy did not employ a cross head, they state that the clearance between the piston and sleeve was such that values of friction measured without the rings were negligible. Essentially then, this apparatus also measured ring friction only.

However, these apparatus were suitable only for measuring piston ring friction under nonfiring conditions. As far as the authors have been able to ascertain, the apparatus described herein is the only one which permits piston and ring friction to be measured in an actual engine under firing conditions.

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OBJECT OF TESTS

The object of the tests reported herewith was twofold, namely:

1. To determine the effect of piston-ring "scuffing" on the piston-friction versus time diagram.

2. To study the piston-friction versus time diagram, particularly during the run-in period, for the following combinations:

Piston types

Aluminum, automotive, and aircraft types

Cylinder sleeve material

Steel, SAE 4140

Piston rings

Automotive and aircraft types

Straight and tapered face

High and low tension

APPARATUS

The engine used by Forbes and Taylor is fully described in reference 1, but for convenience a brief description will be given here. Two sectional views of the engine are shown in figure 1.

The engine consisted of a standard CFR crankcase on which a special cylinder and cylinder head were mounted. The bore of the engine was $3\frac{1}{4}$ inches, the stroke $4\frac{1}{2}$ inches, and the compression ratio 5.05. A shorter connecting rod, 8 inches long, was substituted for the standard 10-inch CFR connecting rod.

The cylinder sleeve (1) was held in position by two annular steel diaphragms (2), clamped to the outer cylinder by means of the cylinder head (5) at the upper end and the steel plate (4) at the lower end.

The cylinder head closed the combustion chamber by means of a piston-shaped section which fitted closely, but did not touch, the cylinder sleeve. A series of grooves machined in this section formed a labyrinth seal (6) which reduced the leakage of gases from the cylinder. This cylinder head will be referred to in this report as "no. 1 cylinder head".

Two spark plug wells (7) and a well containing an optical lever (8) were sealed off from the jacket cooling water by means of flexible neoprene seals (9). Leak-off holes (10) drilled in the side of the cylinder head into the space (12) above the diaphragm took care of gas and oil leakage through the labyrinth and assured atmospheric pressure on the upper diaphragm.

Gas leakage was minimized by pumping oil into the labyrinth through a duct (11) leading through the top of the cylinder head. The oil then passed out the leak-off holes and to the oil reservoir, except for a small amount which leaked into the combustion chamber. The oil was supplied to the head at about 50 pounds per square inch pressure, and was of the same type as the oil used in the crankcase.

A beam of light, thrown on the mirror (8), was reflected onto a photographic film which moved at right angles to the piston motion. Thus, as the sleeve was pulled up and down on the flexible diaphragms by the friction force of the moving piston, a wave form was impressed on the film. The wave form provided a means of determining piston friction under actual operating conditions.

The clearance between the cylinder head and the sleeve was small enough (not more than 0.001 inch on the radius) to prevent any significant loss of pressure in the cylinder. It was very difficult to center the cylinder head in the sleeve with this degree of accuracy, and considerable time was involved in securing the proper alinement. With no. 1 cylinder head it was always necessary to make a check run to determine whether or not the head was touching the sleeve. When contact existed, sticking occurred, and the friction records were not reliable.

In an attempt to eliminate or at least diminish sticking, in the case of actual contact, Forbes and Taylor tried lead-plating the cylinder head. This procedure also allowed considerable control over the clearance. But the results only tended to confirm the impression that if there was any contact whatever between sleeve and cylinder head, the results were unreliable.

As a further precaution against sleeve-sticking, the cylinder head was kept at low temperature, in order to minimize thermal expansion, by circulating tap water through the head independently of the cylinder cooling system.

No. 1 cylinder head was used in the present test program to determine the effect of ring scuffing on piston friction, but at the completion of these runs it was decided to redesign the head in order to overcome its shortcomings.

The difficulty encountered in centering the head with the sleeve was eliminated by constructing a new head embodying a set of junk rings (see figs. 2 and 3). This head will be referred to hereafter as "cylinder head no. 2". The assembly consisted of four continuous (unsplit) rings, A, held in place by four spacers, B, C, D, E, the bottom spacer B serving also as a ring nut for holding the assembly in place. Both rings and spacers were made from mild steel. The junk rings were machined to fit the sleeve with the same radial clearance as the original head, that is, not more than 0.001 inch, but were free to move radially in the spacers a distance of about 0.014 inch. This freedom of radial movement allowed the junk rings to center themselves in the sleeve regardless of any misalinement between head and sleeve. The radial clearance between spacer rings and sleeve was 0.005 inch, which was also more than sufficient to allow for any misalinement between head and sleeve (see fig. 3).

The spacer ring C was grooved circumferentially inside and out, and drilled with eight radial holes. These grooves and holes allowed the passage of an oil stream which was fed into the head through a duct similar to that shown at (11) in figure 1. The oil was supplied at the rate of about 40 drops per minute by means of a Bosch fuel-injection pump, rotated at slow speed by an electric motor. The oil provided a film which helped to close the clearance space between the rings and sleeve, thereby reducing gas pressure leakage as well as providing lubrication.

The oil supply to this new head was critical, and only after considerable experimentation was the correct oil flow determined. When the oil flow was too large, soot and gum formed on the junk rings and in the combustion chamber, and when the oil flow was too small, a large volume of combustion products leaked by, covering the junk rings with carbon. In either case unsatisfactory operation resulted. Aside from soot and gum formation in the combustion chamber, an excess of head oil had the additional disadvantage of changing the characteristics of the oil film on the cylinder walls. In some of the earlier runs, made under conditions of excess head oil feed, an increase in the crankcase oil level was observed after running, showing that some of the head oil had worked its way past the piston and down the cylinder walls to the crankcase. These runs were later repeated using the correct oil feed.

Measurements made on the quantity of oil in the head-oil supply tank before starting a run and after a run, compared with the head-oil leak-off and crankcase contents before and after the run, showed that when the head-oil flow was correctly adjusted, no measurable quantity of oil made its way past the piston to the crankcase, although a small amount was either burned or blown out with the exhaust. Therefore, it is believed that the normal film lubrication characteristics between piston and cylinder sleeve were not upset by the head-oil supply. Additional data to support this belief is presented later.

After the correct usage of this new head and oil system had been determined, excellent performance was realized.

In the original apparatus of reference 1, a film speed of 25 inches per second was used. This meant that, with a reel-capacity of 100 feet of film, continuous records could be taken over a period of 48 seconds only. In order to make it possible to take continuous records over extended periods of time (up to 1 hour with 100 ft of film), a slow-speed film drive (0.295 in. per sec.) was added. By this means it was hoped that any change in friction characteristics could be correlated with operating conditions by comparing the time sequence of the events on the record with the sequence of events entered in the log book.

A short length of a continuous slow-speed record is shown in figure 4.

It should be noted that the slow-speed records show only the maximum amplitude of the friction cycle. This means that these records were useful in showing when a change in the friction cycle took place, but that this change was not necessarily indicative of a change in the work of friction. Thus an increase in the amplitude of the record might be due simply to a more pronounced excitation of the sleeve at its natural frequency rather than to any increase in friction work. This increased amplitude could, of course, be associated with ring scuffing by comparing record, rings and log book after the run.

It was considered desirable, when taking such long slow-speed records of run-in conditions, to have occasional short high-speed "shots" for the purpose of computing friction work. Accordingly, the drive system for the camera was improved to allow for taking high- or low-speed records at will.

The slow-speed motor was connected to the camera drive by means of a chain and sprocket. The sprocket was taken from a bicycle hub and contained a ratchet and pawl system, the pawls being connected to the sprocket and the hub forming the ratchet drum. The ratchet drum was connected to the camera drive.

During slow-speed operation, the pawls engaged the ratchet drum and rotated the camera drive slowly. The high-speed camera motor was connected through a friction clutch directly to the ratchet drum, and when the clutch was engaged the ratchet drum overrode the pawls and turned the camera drive at high speed. The friction clutch on the high-speed drive was engaged by means of a solenoid operated by a switch on the control board.

The position of top center was registered on the high-speed records by means of a neon flash bulb located behind a slit in the camera box and operated by a set of breaker points on the camshaft. These breaker points closed the lamp circuit at approximately top center[1] at the beginning of the power stroke every cycle and impressed a vertical line on the film. Since top center marks were desirable only when taking high-speed records, the lamp circuit was placed in series with the switch which operated the solenoid on the high-speed camera.

One disadvantage of the recording system was that it was always necessary to develop the film before an event could be recognized. In order to overcome this handicap a device for providing a continuous visual record of the friction-time cycle was added to the engine. This device consisted of an electro-magnetic type phonograph pickup, mounted rigidly on the side of the cylinder. The phonograph needle was connected

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[1]For a description of the method of locating top center exactly, see "Discussion".

by means of a thin strip of steel to a blind plug screwed in the spare spark plug hole (see fig. 1). Oscillations of the cylinder sleeve imparted a motion to the phonograph needle, and the output was fed through an integrating circuit and pre-amplifier to a cathode-ray oscillograph where the wave form could be continuously observed. A certain amount of extraneous mechanical vibration was taken up by the pickup unit, and an electrical disturbance was observed at the occurrence of spark, but with a little experience it was an easy matter to correlate this somewhat distorted wave form with the true wave form recorded by the optical system. Another valuable feature of the visual system was that it gave an instantaneous indication as to whether or not the apparatus was working satisfactorily. Lacking this information, a whole run might be wasted. Also the visual system was of assistance in determining the correct oil supply for the new head.

As experience was gained with the visual system and a "feel" was acquired, the need for the 100-foot slow-speed records was practically eliminated. Nevertheless, the slow-speed records were continued for the duration of these tests.

Another addition made to the original apparatus was a gas meter which was connected to the crankcase breather pipe in order to measure blow-by. Precautions were taken to make the crankcase otherwise airtight. A 50-gallon surge tank was inserted in the line between the crankcase breather and the gas meter.

In preliminary work with the friction apparatus, it was noted that the zero, or equilibrium, position of the light spot on the film was not fixed but varied somewhat with operating conditions. It was suspected that this variation might be caused by air or steam bubbles forming in the annular groove in the upper diaphragm, thus preventing uniform contact of the cooling water with all parts of the diaphragm surface. In order to offset any such tendency, two small diametrically opposite holes were drilled into the side of the diaphragm with their axes on the same level as the top of the diaphragm groove. The holes were connected by means of neoprene and glass tubing to the header tank in the cooling system. Although occasional bubbles were observed flowing through the tubes when the engine was firing, no marked decrease in the variation of zero position was noticed.

PROCEDURE

Scuffing Tests

The combination used in the scuffing tests was a cylinder sleeve made from SAE 4140 steel (designated as "sleeve no. 1"), an aluminum piston (designated as "piston no. 2"), and automotive-type cast-iron piston rings (see fig. 5). The piston rings were supplied by the Perfect Circle Company and consisted of standard stock sizes. The ring specifications were as follows:

Groove No.[1]	Manufacturer's designation
1	No. 200, $3\frac{1}{2}$" x $\frac{1}{8}$"
2	No. 70, $3\frac{1}{4}$" x $\frac{1}{8}$"
3	No. 70, $3\frac{1}{4}$" x $\frac{1}{8}$"
4	No. 85, $3\frac{1}{4}$" x $\frac{3}{16}$"
5	No. 85, $3\frac{1}{4}$" x $\frac{1}{8}$"

In order to simplify the procedure, preliminary tests to determine how to create scuffing were made with the engine "motoring", cylinder head removed. A promising way to accomplish this seemed to be to dilute the lubricant on the cylinder walls by introducing kerosene into the cylinder. To this end kerosene was squirted on the cylinder walls while motoring the engine. The attempt failed, however. It was then decided to use ethylene glycol in the same manner, since it is known that ring scuffing results in service engines when this coolant leaks into the cylinders. But this alternative proved no more successful than the preceding one.

It has also been noticed in this laboratory that severe detonation in air-cooled cylinders at high mean-effective pressures (above 200 lb per sq in.) is a contributing cause of ring scuffing; accordingly the cylinder head was replaced,

[1]Piston grooves are numbered from top to bottom.

the engine fired, and severe detonation induced over short periods by means of ethyl nitrite mixed with the inlet air; but no scuffing was produced. The engine was then caused to detonate severely for about an hour by using kerosene as fuel, but without effect, probably because the mean effective pressure of the engine (80 lb per sq in.) was too low.

The cylinder sleeve was then removed, thoroughly cleaned, and placed in a water spray for 44 hours for the purpose of roughening the surface by rusting. The rust did not form uniformly on the surface, but was characterized by long streaks of a soft, powdery consistency. When these streaks were rubbed off, innumerable small pits were found underneath. The roughened sleeve was replaced and the engine fired at 1800 rpm for 2 hours, but with negative results as to scuffing.

Finally the engine was fired while a liberal quantity of ethylene glycol was introduced into the inlet pipe. This procedure produced the desired result, and the record obtained (see fig. 13(i)) showed that the apparatus was indeed sensitive to scuffing. (Compare with record for operation under the same conditions, but without ethylene glycol, fig. 13(h).) The operating conditions for the scuffing runs are summarized in table 1.

Run-In Tests

The remainder of the experimental program was devoted to a study of the friction characteristics of various combinations of aircraft-type piston rings under a fixed set of operating conditions. The piston rings were made especially for these tests by the Perfect Circle Company.

The ring combinations selected were:

- High tension, straight face, compression rings (see fig. 6)
- Low tension, straight face, compression rings (see fig. 7)
- High tension, tapered face, compression rings (see fig. 8)
- Low tension, tapered face, compression rings (see fig. 9)

The same type of scraper rings was used in all these tests (see fig. 10)

All piston rings were factory lapped either to full or line contact, depending on whether they were straight faced or tapered.

A new cylinder sleeve machined from SAE 4140 steel (designated hereafter as "sleeve no. 2") and an aluminum alloy piston (designated hereafter as "piston no. 4") were used in these runs.

The piston (see fig. 11) was similar to the one used in the scuffing tests, having four ring-grooves above the wrist pin and one groove below the pin at the bottom of the piston. The spacing and width of the grooves differed from those of the piston used in the scuffing tests, however. A compression ring was placed in each of the upper three grooves, and scraper rings were placed in the lower two grooves. This arrangement corresponds to current aircraft practice. This piston was a modified version of a design developed by the Ethyl Gasoline Corporation Research Laboratories, who willingly supplied the drawings and castings. The machining of the piston was done by the Perfect Circle Company.

The rings were of cast iron of uniform composition consisting of:

Total carbon	3.50-3.80 percent
Silicon	2.20-3.10 percent
Sulphur	0.10 max. percent
Phosphorous	0.15-0.40 percent
Manganese	0.40-0.80 percent
Molybdenum	0.50-0.70 percent
Chromium	0.20-0.40 percent
Copper	0.50-0.75 (according to section)
Hardness, Rockwell D	40 - 50

The ring tensions, that is, diametral tensions, were measured by means of an instrument especially constructed for this purpose. This instrument is shown in figure 12, and is described in appendix A. By diametral tension is meant that force which when applied across the diameter of the ring at right angles to the gap will be just sufficient to close the gap to that clearance which exists when the ring is in the cylinder, cold.

Measurements of ring tensions and gap clearances, as well as groove clearances, were made before and after each run, but changes in the latter were too small to be significant during the relatively short running periods of these tests.

The lubricating oil used in all runs of this group was SAE 20 (Texaco, aircraft type) obtained from a single barrel which was kept tightly sealed at all times when not in use. The oil barrel was mounted on trunnions and agitated thoroughly before oil was taken from it.

Each run was divided into two parts, designated A and B. Part A consisted of a run of about 1 hour duration during which a continuous 100-foot record was taken at slow film speed (with occasional high-speed shots). This part covered the first phase of the run-in period.

Before starting each 1-hour run, the cylinder sleeve was lapped 50 strokes (25 cycles) with turning motion. An old aluminum piston and cast iron rings, smeared with no. 600 lapping compound, were used for this purpose. The reason for the lapping was to insure that each run would be started with the cylinder sleeve surface as nearly as possible the same in every case. After lapping, the cylinder was thoroughly cleaned by flushing with "Varsol", and a survey of the surface roughness was made with a profilometer. In making this survey the tracer was moved along the surface longitudinally only, that is, in the direction of piston motion. A lacquer specimen of the cylinder surface was then procured by painting a small area of the surface with "Protectol" no. 28 brushing lacquer. This lacquer is quick drying and transparent, and can be readily peeled off the surface in one sheet when dry. The side of the lacquer sheet in contact with the surface carried a presumably true impression of the surface, and thus afforded a ready means of obtaining photographic records of the cylinder surface. All photographs of cylinder surface were made from these replicas. This technique was suggested by the Fuels and Lubricants Division of the Aircraft Engine Research Laboratory of the NACA.

The crankcase was then drained, all oil lines disconnected and blown out with an air hose, oil pump and filter cleaned, and the system flushed thoroughly with kerosene. A small amount of fresh oil was then circulated through the system and drained off, after which the system was filled with the requisite amount of new oil.

The engine was then assembled and a 1-hour run made. The fixed operating conditions were:

Engine speed	1200 rpm
Fuel-air ratio	best power
Spark advance	30°

Manifold pressure	atmospheric
Oil temperature	180° F
Jacket water temperature	180° F
Cylinder head temperature	670-770 F
Inlet mixture temperature	185° F
Oil pressure	44 lb per sq in.
Lubricating oil	SAE 20 (aircraft grade)

After part A the piston and cylinder were removed and measurements taken. The ring surfaces were examined under a microscope for evidences of wear or scuffing. The surface of the cylinder sleeve was surveyed for changes in roughness by means of the profilometer, a lacquer specimen and photomicrograph taken, crankcase oil sampled, and the engine reassembled with the same rings and without lapping the surface. The lubricating oil was not changed.

The engine was then run continuously for 10 hours (pt. B) under the same operating conditions as used in part A, and occasional short records at high film speed were taken.

At the end of the 10-hour run the cylinder and piston rings were again inspected and measurements, similar to those made after the 1-hour run, were recorded.

This procedure was followed for each of the four combinations of rings listed before. Condensed data on the runs are given in tables 2 to 5. A summary of the run-in test program follows.

Before starting run-in tests

- Typical surface of unused piston rings photographed, and surface roughness measured with a profilometer, in the circumferential direction
- Lubricating oil sampled

Prior to each run

- Oil system flushed and refilled with fresh oil (prior to 1-hr runs only)
- Cylinder lapped (prior to 1-hr runs only)
- Cylinder cleaned with Varsol
- Lacquer specimen of cylinder surface obtained and photomicrographed
- Measurements:
  - Cylinder surface finish (profilometer)
  - Piston ring tension - all rings
  - Piston ring clearance in each groove (feeler gage)
  - Piston ring gap in standard gage

During runs

All usual readings taken.
Continuous film and/or occasional high film-speed cycles taken.
Performance checked with cathode-ray oscillograph
Blow-by measured.

Test conditions

Engine speed	1200 rpm
Fuel-air ratio	best power
Spark advance	30°
Manifold pressure	atmospheric
Oil temperature	130° F
Jacket water temperature	180° F
Cylinder head temperature	67°-77° F
Inlet mixture temperature	185° F
Oil pressure	44 lb per sq.in.
Duration	1 to 10 hours

After each run

Lacquer specimen of cylinder surface obtained and microphotographed.
Crankcase oil sampled
Measurements:
- Cylinder surface finish (profilometer)
- Piston ring tension - all rings
- Piston ring clearance in each groove (feeler gage)
- Piston ring gap in standard gage

Foregoing procedure followed on aircraft-type piston rings for two extremes

High tension, straight face, cast iron rings - - - - runs 13A and 13B
Low tension, straight face, cast iron rings - - - - runs 14A and 14B
High tension, tapered face, cast iron rings - - - - runs 15A and 15B
Low tension, tapered face, cast iron rings - - - - runs 16A and 16B

After completion of run-in tests

Typical surface of used piston rings photographed and surface roughness measured with a profilometer.

DISCUSSION

Interpretation of Friction Records

For the purpose of illustration, consider the friction record shown in figure 14(a). This trace really represents a record of cylinder-sleeve displacement versus time. Three consecutive engine cycles (12 strokes) are shown in this record. The order of events is from left to right and the 4 strokes in the first cycle (namely, power, exhaust, suction, and compression) are indicated by the letters P, E, S, and C, respectively. The duration of each stroke is indicated by the vertical lines drawn across the trace. These lines were drawn on the records with ink after developing them and the spacing was determined by dividing the time duration of 1 engine cycle into 4 equal parts. The first vertical line, farthest to the left at the beginning of the power stroke, represents true top center and was drawn in with ink. The exact location of this line was determined from the second vertical line (the long one) which was impressed on the film by the neon flash bulb in the camera. Although this neon bulb was made to flash exactly when the piston reached top center at the beginning of the power stroke, it did not necessarily indicate the true position of top center on the film trace because the image of the light spot reflected from the oscillating mirror on the cylinder sleeve was often displaced slightly to the right or the left of the axis of the camera lens. Since it was somewhat of a nuisance to adjust the optical system to overcome this spot displacement every time the engine was taken apart and reassembled - which was often - the displacement or "top center error" was allowed to persist and a photographic check was made to determine its magnitude. This was done for each record by taking an image of the light spot and then flashing the neon bulb while holding the film stationary. These check images are not shown in the figures. The top center error varied slightly after each engine overhaul.

The records shown in this report, with one or two exceptions, have all been more or less traced for the purpose of reproduction. The published records, therefore, exhibit a roughness which was nonexistent in the original negatives and most of the fine detail has been lost. The tracing was necessary because the brightness of the reflected light beam was very often diminished by condensate forming on the inside of the lens in the mirror well (8) (fig. 1). This was due to small amounts of water vapor leaking past the neoprene seal (9). Although this seal was satisfactory as far as actual water leakage was concerned, it was difficult to prevent minute quantities of vapor from seeping through.

It will be noticed that at the beginning of each stroke, the trace rises or falls abruptly. A rising of the trace indicates that the sleeve is moving upward and vice versa.

The greatest deflection of the trace occurs at the beginning of the power stroke where it will be observed that the trace falls abruptly, then rises with equal abruptness, then falls again, and finally rises slowly for the remainder of the stroke. Although this fluctuation at the beginning of the stroke actually represents the motion of the sleeve, it is not representative of the friction force acting on the sleeve because the initial excitation has caused the sleeve to oscillate at its natural frequency.

The instantaneous friction force acting would be represented by a mean curve drawn through these oscillations. In this report no attempt was made to measure instantaneous friction; all conclusions were drawn on the basis of friction work for the cycle. In computing friction work for the cycle, all effects of natural frequency cancel out provided the work loop is closed at both ends.

At the beginning of the exhaust stroke the sleeve is pushed abruptly upwards, but the natural frequency is not so prominent. By comparing this effect with that at the beginning of the power stroke, the effect of pressure behind the piston rings becomes immediately evident.

Also at the beginning of the suction stroke the sleeve is pulled abruptly downward, and at the beginning of the compression stroke it is pushed abruptly upward. The magnitude of the displacement is about the same in either case and less than that of the power and exhaust strokes.

These phenomena are repeated with excellent cyclic regularity and hence, in the runs which follow, only single cycles are shown.

Comparison of Friction Work for Scuffing Runs

Records of piston friction taken during the scuffing runs are shown in figure 13, operating conditions are given in table 1, and friction work is summarized in table 6. The method used to compute friction work is given in appendix B. All data presented here may vary from true values by a maximum of ±4 percent (see sec. under Precision).

Runs 3 and 4 were made under conditions of normal combustion and severe detonation respectively. The friction records are shown in figures 13(a) and 13(b). The timing marks for these two runs were recorded during the exhaust strokes, but in all other runs the timing marks were recorded during the power strokes.

It is evident that an alteration in the friction wave form accompanies the change in combustion. This alteration occurs at the beginning of the power stroke, as would be expected, and is recognized as a more pronounced excitation of the sleeve at its natural frequency. The friction work for these two records is the same, however.

Records for the motoring and firing runs (nos. 5 and 6) made after the sleeve had been freshly rusted are shown in figures 13(c) and 13(d). These runs show a considerable increase in the friction work over that of runs 3 and 4. Since these four runs were all made under the same operating conditions (except for inlet temperature, which was about 40° F higher in the first two runs), the increase is apparently due to the rusted cylinder surface.

Also, the friction work for the motoring run, no. 5, is less than that for the firing run, no. 6, as would be expected. The next firing run, no. 7, was made under the same conditions as run no. 6, although at a later date, and it will be noticed that the friction work is somewhat less. This difference is probably due to the fact that the sleeve, which was freshly rusted in run 6, had been thoroughly run-in by the time run 7 was made, with a consequent reduction in friction. The record for run 7 is shown in figure 13(e).

The same conclusion can be drawn as regards runs 5 and 8, which are motoring runs made under the same conditions. Run 5 was made when the sleeve had been freshly rusted, and run 8 was made at a later date when the sleeve had been subjected to several hours' running. Thus it would be expected that the friction work would be less in run 8. The record for run 8 is shown in figure 13(f).

Run 9 was a continuation of the motoring run no. 8, and 180 cubic centimeters of ethylene glycol was added to the incoming mixture in an attempt to make the piston rings scuff. The record, taken immediately after the ethylene glycol had been added, is shown in figure 13(g). There is no marked change in the wave form of figure 13(g) compared to that of figure 13(f), and it was accordingly concluded that the rings did not scuff after the addition of the ethylene glycol under motoring conditions. The friction work for run 9 is slightly larger than that for run 8.

The ignition was then turned on immediately after the motoring run to see if the ethylene glycol which had been previously added would cause the rings to scuff under firing conditions. The friction work for this run, no. 10, is about the same as that for the firing run, no. 7, in which the cylinder wall lubrication was uncontaminated.

The record obtained during run 10 is shown in figure 13(h). Notice that the record for run 7 (see fig. 13(e)) is almost identical with that for run 10, except that in run 10 the natural frequency of the sleeve is more pronounced. From a comparison of the records, it was concluded that no scuffing took place during run 10.

Figure 13(i) shows the record obtained while 300 cubic centimeters of ethylene glycol was added to the inlet air with the engine firing (run 11). A radical change in the wave form is apparent, and it will be observed that the trace is no longer exactly periodic. This film records the result of scuffing. When the piston rings were removed after this run, they were found to be scuffed. The friction work for run 11, given in table 6, has two values, a and b, corresponding to the work for each of 2 cycles in which the wave form was at variance.

It is rather surprising to notice that the friction work during scuffing is not greater than during normal operation (compare runs 7, 10, and 11). Apparently the friction work with or without scuffing is about the same.

Comparison of Friction Work for Run-In Tests

The friction work associated with the various piston ring combinations used in the run-in tests, runs 13 to 16, is given in table 7, and the friction records are shown in figures 14 to 17.

Figure 14(a) shows the record taken 6 minutes after the start of run 13A when the rings (high-tension, straight face) were new. The record taken 1 hour and 5 minutes after the start of this run is shown at (b). The displacements at the beginning of each stroke are now less pronounced although the pattern of the cycle is similar to that shown at (a). The friction work has decreased by about 14 percent.

After the completion of run 13A the cylinder, piston, and rings were removed and cleaned, then reassembled, and run 13B commenced. The record shown at (c) was taken 32 minutes after the start of run 13B and is very similar to (b) except that the natural frequency effects at the beginning of the strokes have still further diminished. The friction work for this record is only about $1\frac{1}{2}$ percent less than that of (b).

Ten hours after the start of run 13B the friction record has been taken on the appearance shown at (d). A very substantial increase in the natural frequency oscillation at the beginning of the power stroke has taken place while such oscillations at the beginning of the other strokes of the cycle have practically disappeared. This phenomenon is not easy to explain.

A glance at the friction records for the other ring combinations used in the run-in tests (figs. 15 to 17) reveals that these records all follow the same general pattern displayed by the records for runs 13A and 13B. Also as in the case of runs 13A and 13B the natural frequency excitation does not seem to be a function of running time. In the record shown in figure 17(a) the natural frequency excitation at the start of the run is practically nonexistent, whereas after more than 10 hours' running, it is very much in evidence (fig. 17(d)). Apparently the only generalization that can be made regarding this effect is that, for any cycle, the effect is greatest at the beginning of the power stroke.

The data given in table 7 show that during the run-in period:

(a) The work of friction decreases steadily with running time, the most rapid decrease occurring with high-tension, tapered face rings.

(b) The decrease in friction work with running time is not due to a decrease in ring tension, which remains essentially constant.

(c) In the case of straight-face rings, friction work is no greater for high-tension than for low-tension rings.

(d) In the case of tapered-face rings, friction work is greater for high-tension than for low-tension rings.

(c) At the end of the run-in period (about 11 hours), tapered-face rings give smaller values of friction work than straight-face rings.

Reduction in friction after cleaning parts. - It is of interest to notice that the friction work at the end of the 1-hour run (part A) is always greater than the friction work at the beginning of the 10-hour run (part B) (see table 7). It will be remembered that after the 1-hour run the cylinder, piston and rings were removed and measurements taken on surfaces, ring tensions, and so forth. The engine was then reassembled with the same rings, piston, and cylinder, and the 10-hour run commenced. The cylinder surface was not lapped between the 1- and 10-hour runs[1] nor was any other change made except to clean the rings, piston, and cylinder before reassembling.

The cleaning process consisted merely of flushing the parts in unleaded gasoline since they were always in excellent condition after only an hour or so of running.

If the piston rings were inadvertently stretched sufficiently to change their diametral tensions while removing or replacing them on the piston, then some change in friction work might be expected, although this should tend to increase the work. However, great care was used in placing and removing the piston rings, and repeated checks made on specimen rings showed that no significant change in diametral tension resulted from this procedure. Moreover, since at least 15 minutes was allowed at the beginning of the 10-hour run before taking the first record, and lubricating oil and jacket water were approximately at the standard values before starting the engine, it would be expected that the oil film on the cylinder walls would be normal.

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[1]Except in the case of run 14B. Before starting this run the cylinder was lapped in order to remove a small trace of rust which formed on the cylinder wall overnight. The rusting was due to water vapor which condensed in the cylinder head leak-off passages during the preceding run but was in no way serious. The rust might well have been removed by wiping with a cloth, but it was considered safer to remove it by lapping. After this occurrence the cylinder head was always removed immediately after completing a run. The effect of this lapping, however, does not in any way modify the discussion which follows.

Hence there does not seem to be any obvious reason why this consistent reduction in friction work occurred after the rings, piston, and cylinder had been removed and cleaned.

Effect of excess cylinder lubrication on friction work. - In the first two runs made in this series an excess of oil was supplied to the ring assembly on the cylinder head, and some of this oil worked its way past the piston to the crankcase. Although these runs were rejected when this was discovered, it was later decided to compute the friction work for the runs in order to appraise the effect of the excess lubrication on the cylinder walls. The runs (shown in fig. 18 and designated x and y, table 7) were made with the same ring combination as used in runs 13A and 13B (straight face, high tension), but notice that the friction work is much less than that for runs 13A and 13B. This fact lends weight to the general assumption that the piston and rings operate under conditions of partial film lubrication. Otherwise an excess of lubrication would not reduce the friction.

Also notice that in the case of run x (and allowing for a ±4 percent cyclic variation) the friction work does not change significantly with time, while in the case of run y a definite increase may be recognized. In run x a constant head-oil flow of 13 cubic centimeters per minute was maintained, and this same value was used at the start of run y, but during the latter run the flow was cut down so that when the last record was taken, only 5 cubic centimeters per minute were supplied to the head. This reduced flow probably brought the cylinder walls closer to the normal condition of lubrication with a resulting increas in friction work.

These results further confirm the assumption that under normal conditions the piston and rings operate in the partial film region.

It may also be observed, as in the case of all the other runs in this group, that after the rings, piston and sleeve were removed and cleaned, there was a reduction in the friction work.

Ring tensions. - Ring tensions of piston rings, used in the run-in tests, measured before and after the runs, are given in tables 2 to 5. The changes in ring tensions were not great, but an actual increase in ring tensions with running time was observed in many cases. In other cases a small decrease in tension was observed with time. There does not appear to be a correlation between the type of ring and the change in tension. The change was probably due mostly to the effect of cylinder temperature on internal stresses in the ring rather than to any effects of wear - at least for the short running periods here considered.

Gap clearances. - Gap clearances showed an increase with running time in all cases.

The change in gap clearances for all rings was approximately the same during the 1-hour run as during the 10-hour run showing that the apparent rate of wear was approximately ten times as great during the first hour of running. High and low tension, straight and tapered face rings all showed about the same increase with running time, but the beveled scrapers showed considerably greater increases.

Blow-by. - In the case of straight face rings, blow-by was greater for low tension rings than for high tension rings during the first hour of running.

In the case of tapered face rings, this trend was reversed during the first hour of running.

Also during this period the straight face rings gave greater values of blow-by than the tapered face rings.

These deductions may not be general because there was considerable scatter of the plotted values of blow-by, and the 1-hour runs allowed time for the taking of only two or three measurements.

During each 10-hour run many observations of blow-by were taken (usually ten to twenty), although only a few of these data are shown in tables 2 to 5. However, all values of blow-by for the 10-hour runs are plotted in figure 19, where it will be observed that the scatter is large and the trends erratic.

It does not seem advisable to draw conclusions on the basis of these curves because it appears as though some unforeseen circumstance, such as the lining-up of ring gaps due to a slow rotation of the rings, was operating to influence the results.

Surface Characteristics of Piston Rings

Used in Scuffing Runs

Photomicrographs of the automotive-type piston ring surfaces are shown in figure 20. Figure 20(a) shows the surface of a new cast iron compression ring of the type used in the scuffing runs. The large horizontal dark lines represent tool marks which on the actual ring are just visible to the naked eye. The light, vertical lines represent scratches which cannot be seen with the naked eye.

Figure 20(b) is a photomicrograph of a similar ring after many hours of normal operation. Note that the tool marks and scratches have almost entirely disappeared. The actual ring surface has a shiny, polished appearance.

Figure 20(c) shows a ring, of the same type as the other two, which was scuffed in run 11. Note the deep gouges and ragged edges.

Surface Characteristics of Cylinder Sleeve

Used in Run-In Tests

Photomicrographs of the cylinder surface replica taken after lapping are shown in figure 21. The lapping marks are clearly visible as the oblique crisscross lines extending from the top to the bottom of the pictures. These lines are the result of turning the lapping piston while moving it in and out of the cylinder. In figure 21(a), which shows the lapped surface before run 13A, several heavy, horizontal, dark lines may be observed. These lines are probably grinding marks as the sleeve was new at the beginning of run 13A. The lines do not appear in any of the other photographs showing, probably, that they have been worn off.

A comparison of these photographs indicates that lapping produced a surface of reasonable uniformity, although the surface shown at (c) seems to be somewhat coarser than the others. The profilometer, however, indicated that the lapped surface shown at (c) was actually smoother (10 microinches average) than the other lapped surfaces (15 microinches average).

This disagreement between visual comparison and the profilometer readings may be due to photographic technique or to the method of averaging the profilometer readings. The profilometer readings varied over a range of about 5 rms microinches for these surfaces, and a mental average of this variation was taken.

Figure 22 shows the lapped surfaces after the engine was run for about an hour and a quarter, in each case, with different piston ring combinations.

The oblique crisscross lines are fainter in these photographs showing that the lapping marks have been worn off by the piston motion. The scratches made by the piston and rings now predominate and can be recognized as the series of heavy vertical lines. The horizontal, dark lines can still be seen at (a) although they are fewer and less distinct.

The profilometer indicated that the lapped surfaces were worn smoother by running except in the case of run 15A. The average increase in smoothness was about 3 microinches.

No correlation has been established as yet between profilometer readings and cylinder wear. Moreover the precision of surface measurements in this work was poor.

Figure 23 shows the condition of the same surfaces[1] after each has been subjected to about 10 hours' additional running. There does not appear to be any great difference between the surface after a 1-hour run as compared with a 10-hour run. This observation would seem to indicate that, after lapping, the surface approaches its ultimate condition in about an hour's running time. This observation also agrees with the measurements made on piston-ring gap clearances.

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[1]As noted previously the surface was lapped before starting run 15B in order to remove a trace of rust. The lapping does not appear to have affected the condition of the surface at the end of this 10-hour run.

Surface Characteristics of Piston Rings

Used in Run-In Tests

Photomicrographs of new aircraft-type piston ring surfaces are shown in figure 24. These rings are all factory-lapped. The surface of a straight-face compression ring is shown at (a); the vertical lapping marks cover the entire surface showing that it has been lapped to full contact. This surface registers 25 microinches rms on the profilometer.

The bearing surface of a beveled scraper ring is shown at (b). This ring is also lapped to full contact.

A tapered-face ring lapped to "line contact" is shown at (c). The vertical lapping marks are visible along the lower edge of the ring.

These photographs show that the new rings all have bearing surfaces of very nearly the same character. These lapping marks are not visible to the naked eye; the surfaces have a uniform dull appearance.

Photographs of the same ring surfaces after about 11 or 12 hours' use are shown in figure 25. The lapping marks are still clearly visible although they have been rubbed down considerably. The straight-face compression ring at (a) now registers 14 microinches on the profilometer. It is rather interesting to notice the series of fine horizontal lines on the surface of this straight-face ring. No such lines appear in figure 24(a), the photograph of the same surface before use, and apparently there is no manner in which these lines could be produced by piston motion. It would seem, therefore, that these lines must be tool marks which were obscured in the photograph of the original surface by the depth of the lapping marks, but which reappear as the ridges formed by lapping wear down. Similar horizontal lines may be observed on the upper part of the used tapered-face ring shown in figure 25(c), but these lines do not show on the upper part of the unused tapered-face ring shown in figure 24(c). If these lines were really tool marks, they would be expected to show on the upper part (unlapped region) of this surface. Perhaps it is possible that these tool marks (if such they are) do not show up until they have had carbon or oil rubbed into them while running.

Photomicrographs of Piston Rings

Photomicrographs of specimen piston rings are shown in figure 26. Figure 26(a) shows the unetched surface of an automotive piston ring of the type used in the scuffing runs. The black marks show the graphite, inclusions, and so forth.

An etched photomicrograph of the same ring is shown in (b). The dark stringers and spots represent graphite, the dead white areas represent carbide, and the background material is perlite. The carbide areas are large and spotty.

Figure 26(c) shows the unetched surface of an aircraft ring of the type used in the run-in tests. The graphite, inclusions, and so forth seem to be more pronounced in this ring.

Figure 26(d) shows a photomicrograph of the aircraft type ring surface etched. The large gray areas represent graphite, and the narrow white areas represent carbide. The carbide areas are small and well distributed. The background white represents ferrite which is evidently more evenly distributed and present in larger proportion than in the automotive ring.

PRECISION

Cyclic variation in friction work. - Three consecutive friction cycles were selected from the record taken at the beginning of run 13A, and three consecutive cycles were selected from the record taken at the end of run 13A. Run 13A was a 1-hour run.

Also, 3 consecutive friction cycles were selected from each of the records taken at the beginning and end of run 13B. Run 13B was a 10-hour run. These records are shown in figure 14.

The variation of the friction work from cycle to cycle, and the percentage variation of each cycle from the average of 3 cycles, is given below.

	Cycle no.	Friction work area (sq. in.)	Percentage variation from average
Record taken at beginning of Run 13A (1-hour run)	1	9.35	+3.6
	2	8.60	-4.6
	3	9.11	+0.8
		Av. = 9.02	
Record taken at end of Run 13A (1-hour run)	1	7.68	+1.5
	2	7.41	-2.1
	3	7.62	+0.6
		Av. = 7.57	
Record taken at beginning of Run 13B (10-hour run)	1	7.75	+2.1
	2	7.31	-3.7
	3	7.71	+1.5
		Av. = 7.59	
Record taken at end of Run 13B (10-hour run)	1	6.01	-0.1
	2	6.17	+2.1
	3	5.92	-2.0
		Av. = 6.03	

The foregoing data show that the percentage variation in friction work from cycle to cycle is about ±4 percent. Since the run and the cycles selected represented random choices, it is probably safe to assume that this variation is representative of all runs. Therefore, comparisons of friction work made on the basis of single cycles should be valid to within ±4 percent.

Although the precision could have been substantially improved by making comparisons on the basis of several cycles, all comparisons made in this work were based on single cycles, because the work involved in enlarging, tracing, and transforming the records from their original coordinates to coordinates suitable for computing the friction work was considerable (see appendix B).

Reproducibility of results. - After the completion of runs 13A and 13B (high-tension, straight-face rings), it was decided to repeat run 13A by way of determining how closely the results could be duplicated. A new set of piston rings, having the same diametral tensions, were selected, and the cylinder was lapped, oil system flushed, and so forth, so that at the start of the check run the apparatus was very nearly in the same condition as at the start of run 13A.

The diametral tensions of the piston rings measured before the runs were:

	Ring number				
	1	2	3	4	5
Run 13A	5.70	5.85	5.86	9.90	10.05 lbs
Check run	5.72	5.85	5.87	9.86	10.00 lbs

Unfortunately the record taken at the beginning of the check run was lost due to a break in the film, but a good record was obtained at the end of the run. This record is shown in figure 27 and compares favorably with the record shown in figure 14(b).

The friction work of each of the 3 consecutive cycles shown in figure 27 was computed and compared with the work for each of the cycles shown in figure 14(b). Results appear in the following table:

	Cycle no.	Friction work area (sq. in.)
Record taken at end of run 13A	1	7.68
	2	7.41
	3	7.62
		Av. = 7.57
Record taken at end of check run	1	7.26
	2	7.42
	3	7.60
		Av. = 7.43

The average friction work for the check run is 1.9 percent less than the average friction work of run 13A. The apparatus, therefore, yields reproducible results.

Calibration errors. - The over-all sensitivity of the apparatus, that is,

$$\frac{\text{force acting on cylinder sleeve, pounds}}{\text{vertical deflection of film trace, inches}}$$

was determined by removing the cylinder head and loading the sleeve-diaphragm system with weights while recording the displacement of the light spot on a stationary film.

During the calibration the jacket water was maintained at 180° F because this was the standard jacket temperature for all runs. What change, if any, took place in the over-all sensitivity of the apparatus under firing conditions is unknown, but such change was probably small since the diaphragms were in no way exposed to cylinder temperatures, and, moreover, the diaphragms formed the upper and lower surfaces of the water jacket (see fig. 1) through which a good circulation was maintained.

The scuffing runs were performed with sleeve-diaphragm system no. 1, and the total running time for these runs was about 6 hours. A calibration of this sleeve-diaphragm system was made immediately after these runs, and since the diaphragms were thoroughly "seasoned," having been used at least 100 hours in preliminary runs, it was assumed that no significant change in the calibration took place during the brief period of the scuffing runs.

The calibration curve for sleeve-diagram system no. 1 is shown in figure 28.

In the case of sleeve-diagram system no. 2, which was used in the run-in tests, the experimental work occupied a period of about 4 months and involved well over 100 hours' running time. A calibration curve was determined before the runs, when the sleeve and diaphragms were new, and another curve was determined after the runs were completed. A maximum change of 5 percent in the over-all sensitivity was noted. An average value of over-all sensitivity was therefore assumed, giving a calibration error of $\pm 2\frac{1}{2}$ percent. The calibration curves are shown in figure 29.

Personal errors. - Errors made in enlarging the records in a projector and tracing the enlarged image manually for the purpose of constructing work loops should, by the nature of the process, cancel out.

Film speed. - The high-speed film drive was powered by a 1/8-horsepower synchronous motor which brought the film drum up to speed in a fraction of a revolution, but as a precaution, the first few cycles of a record were discarded. Repeated use of this camera in many investigations in this laboratory have shown that it is thoroughly reliable and free from any significant errors.

Blow-by measurements. - Blow-by was measured by a new precision bellows-type gas meter which gave readings within ±1/2 percent when calibrated at the factory. A volume of 1 cubic foot was indicated by 1 revolution of a pointer on a large dial on the front of the meter. Although it would have been desirable to take readings based on 2 or 3 revolutions of the pointer, this procedure could not be followed especially in the case of the 1-hour runs, because the blow-by was so small that the better part of an hour was required in some cases for the pointer to complete 1 revolution. Consequently, since blow-by measurements were desired at the beginning and end of a 1-hour run, readings were taken over 1/2 revolution of the pointer only, but always over the same arc.

Ring tensions. - Diametral ring tensions were measured with a precision of ±1 percent (see appendix A).

Ring gap clearance. - Gap clearances in piston rings were measured by means of a feeler gage to the nearest 0.001 inch with an error of approximately ±3 percent.

Over-all precision of experimental work. - In view of the foregoing considerations it is believed that an over-all precision of ±4 percent is a reliable figure to use in interpreting data presented herein.

CONCLUSIONS

1. Neither scuffing nor severe detonation at moderate mean effective pressure (80 lb per sq in.) increases the friction work.[1]

2. Motoring friction work is less than firing friction work.

[1]In these conclusions "friction work" means that work due to piston and ring friction only.

3. The rate of wear on cylinder barrels, as measured by profilometer readings, and the rate of wear on rings as measured by gap increases is much greater during the first hour of running than during the remaining 10 hours of an 11-hour running period.[a]

4. During an 11-hour period and within the range of variables tested:

(a) The work of friction decreases steadily with running time, the most rapid decrease occurring with high-tension, tapered-face rings.

(b) The decrease in friction work with running time is not due to a decrease in ring tension, which remains essentially constant.

(c) The decrease in friction work with running time is undoubtedly due to the change in surface characteristics of cylinder walls and rings. Lapped cylinder-wall roughness fell from 15 microinches to 10 microinches and lapped-ring roughness fell from 25 to 14 microinches.

(d) In the case of straight-face rings, friction work is no greater for high-tension than for low-tension rings.

(e) In the case of tapered-face rings, friction work is greater for high-tension than for low-tension rings.

(f) At the end of the run-in period (11 hours) tapered-face rings give smaller values of friction work than straight-face rings.

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[a]No correlation has been established as yet between profilometer readings and cylinder wear. Moreover the precision of surface measurements in this work was poor; also the NACA has found that changes in gap clearances do not correlate well with weight changes of piston rings and therefore do not give a reliable indication of wear. The differences in the rates mentioned above, however, are of the order of 5 to 1, and should therefore be of some significance.

5. Photomicrographs of the structure of commercial cast iron rings show that the material customarily used in aircraft rings

(a) shows much larger percentage of ferrite

(b) shows more free carbon

than the material customarily used for automotive rings.

Massachusetts Institute of Technology,
Cambridge, Mass., January 27, 1944.

APPENDIX A

DESCRIPTION OF APPARATUS FOR MEASURING

DIAMETRAL TENSION OF PISTON RINGS

A photograph of the apparatus used to measure diametral ring tension is shown in figure 12.

The piston ring was placed between the adjustable shoulder A and the sliding arm B. The sliding arm was connected by means of two pieces of steel wire C to the upper end of a spring balance D. To the lower end of the spring balance, a screw E was connected which, when tightened, transmitted a load to the piston ring, through the balance, where its magnitude could be read directly.

The correct gap clearance of the ring was determined by placing the ring in a ring gage, machined to the exact size of the cylinder bore, and measuring the clearance with a feeler gage.

The ring was then placed in the apparatus and the micrometer screw F tightened until the ring was compressed to the correct gap clearance. When the micrometer screw was in contact with the vertical shaft G on the sliding arm, a circuit was closed which caused the small lamp H to light.

The screw E was then tightened until the lamp went out, which indicated that the contact between the micrometer screw and shaft G was broken. At this point the load on the ring was equal to the ring tension, and the magnitude was read on the spring balance.

The plate I was used to line up the ring accurately. The height of the plate was just equal to the radius of the cylinder. When the ring was placed in the apparatus, care was taken to have the gap in line with the top of the plate. The plate also served to line up the ring parallel to the base of the machine. After lining up the ring, the plate was removed.

The apparatus was flexibly mounted in a vertical position, and a small motor J, with an unbalanced disk on the shaft, was used to vibrate the apparatus in order to diminish static friction. The motor was mounted on the back of the apparatus, at the top, and does not show to advantage.

Repeated measurements on a given ring were in agreement within one percent. All values read off the spring balance were corrected for the weight of the sliding arm and those parts hanging on the sliding arm.

APPENDIX B

COMPUTATION OF FRICTION WORK

In computing friction work from the film records it was first necessary to determine the deflecting force acting on the cylinder sleeve, corresponding to a given ordinate on the film record. This information was obtained as explained in the section on Precision.

The calibration curves are shown in figures 28 and 29.

In the following exposition, only the data pertaining to sleeve-diaphragm system no. 2 will be used.

The over-all sensitivity of the apparatus is defined as

$$K = \frac{F}{D_f} = \frac{\text{force acting on cylinder sleeve, pounds}}{\text{vertical deflection of film trace, inches}}$$

From figure 29 the average value of K is 390 pounds per inch.

It is necessary to convert the time scale on the film to a distance scale so that the product of the two coordinates will have the dimensions of work. The records shown in this report give the variation of sleeve displacement with time during the engine cycle.

If the time axis (which, at constant film and engine speed, is directly proportional to crank angle) is tranformed to give the distance of the piston from top center at any instant, then the summation of the product of the two coordinates is proportional to the friction work.

The time scale on the film can be converted into degrees crank angle and the corresponding distance of the piston from top center determined by straight-forward measurement and computation, but the process is laborious. Considerable effort was saved by using the M.I.T. transfer machine. The transfer machine is an apparatus used in transforming pressure-crank angle indicator diagrams to pressure-volume diagrams.

A photograph and schematic diagram of the machine are shown in figures 30 and 31.

The machine consists essentially of two sliding tables A and B which are free to move in a horizontal direction. The table A is actuated by the belt C which passes over the drum D and the pulley E. The table B is actuated by the disk F and the connecting rod G. The drum and the disk are rigidly fastened to the shaft H. When this shaft is rotated through a given arc, the linear displacement of the table A is directly proportional to the arc, while the linear displacement of the table B is directly proportional to the linear displacement of a piston in an actual engine having the same ratio of crank radius to connecting rod length. This ratio may be varied on the transfer machine by changing the effective length of the rod G by means of the slot J and the clamp at K.

A pressure-crank angle diagram is placed on table A, and a blank sheet of paper is placed on table B. The "top center" position for both tables is fixed by placing the pointer L on the top center line on the pressure-crank angle diagram and moving the table B as far to the right as possible. A clamp M on table A allows this table to be moved independently of the belt C while making this adjustment.

After the tables have been lined up on top center, the drum D is rotated by hand until the pointer L is at bottom center position on the pressure-crank angle diagram. The pointer L is connected to the stylus N through the rod P which is constrained to move transversely by the guides Q.

A small electric motor R which drives drum D at very slow speed is then started. As table A moves to the left, the pointer L is held on the indicator line by the operator. The stylus N then traces out the corresponding pressure-volume diagram.

In this transfer process 180° on the crank-angle scale is transferred to a scale 5 inches long on which the distance of the piston from top center is represented. In the case of indicator diagrams, this scale must be multiplied by the area of the piston to complete the pressure-volume transformation, but in the case of friction records, cylinder volumes are not necessary, and the transferred scale can be used directly.

The transfer machine accommodates a pressure-crank angle indicator diagram which is 9 inches long for 360° of crank rotation. Thus, in using the machine to transfer friction records, each 360° interval must be enlarged to 9 inches. This enlargement was effected by placing the record in a photographic enlarger and tracing the image on cross-section paper with a pencil.

One engine revolution on the film record is represented by

$$l = \frac{60\ V_f}{N} \text{ inches}$$

where

V_f film speed, inches per second

N engine rpm

The required enlargement of the film record is then

$$M = \frac{9}{l} = \frac{9\ N}{60\ V_f}$$

In enlarging the abscissa scale on the film record the ordinate will be correspondingly enlarged so that 1 inch on the enlarged ordinate scale will represent K/M pounds per inch. The ordinate scale is not altered on the transfer machine, however.

Two consecutive strokes (1 revolution) of the enlarged friction record are then placed on the transfer machine and converted into a force-piston-position diagram. The result is shown in figure 32. The upper diagram represents the loop obtained if the two consecutive strokes happen to be the suction and compression strokes and the lower diagram represents the result obtained if the power and exhaust strokes are taken.

Since the stroke of the engine is 4.5 inches, 1 inch on the transformed abscissa scale will represent 4.5/5 inches of piston motion. Hence 1 square inch of the friction work loop represents

$$\frac{K}{M} \times \frac{4.5}{5} \times \frac{1}{12} \text{ foot pounds}$$

If A_1 represents the area of the friction work loop for the power and exhaust strokes and A_2 represents the area of the friction work loop for the suction and compression strokes, then the friction work for 1 cycle is

$$W = (A_1 + A_2) \times \frac{K}{M} \times \frac{4.5}{5} \times \frac{1}{12}$$

and with $K = 390$ pounds per inch for the case of sleeve-diaphragm system no. 2 and a film speed V_f of 25 inches per second, this expression becomes

$$W = \frac{4870}{N} (A_1 + A_2) \text{ ft-lbs per cycle}$$

The piston friction mean effective pressure is defined as

$$\text{pfmep} = \frac{\text{friction work of cycle, inch-pounds}}{\text{piston displacement, cubic inches}}$$

and is given by

$$\text{pfmep} = \frac{4870}{N} (A_1 + A_2) \times 12 \times \frac{1}{37.3}$$

$$= \frac{1570 (A_1 + A_2)}{N} \text{ pounds per square inch}$$

where the figure 37.3 is the piston displacement in cubic inches, and the factor 12 converts foot-pounds to inch-pounds

Piston friction horsepower is given by

$$\text{pfhp} = \frac{W \times N/2}{33{,}000} = 0.0737 (A_1 + A_2)$$

It may be pointed out that the work loops shown in figure 32 are almost, but not completely, closed. This situation obtains to a greater or lesser degree for all records. The beginning and end of the friction trace for a given stroke is indicated by little circles at the right and left hand ends of the loops. The right hand end of the loops is always closed, of course, because the trace has not been broken here, but when the lower branch of the loop is folded backward, it does not always meet the beginning of the upper branch. This is due to resonant excitation of the sleeve, and gives a slightly erroneous value for the friction work of 1 revolution. In the case of figure 32 the work areas for both loops would be greater than the true work area.

In other cases, however, the two branches of the loop cross at the left hand side, giving a smaller value of friction work than the true value.

On the average there were about as many loops which gave high values of friction work as there were loops giving low values, and the maximum variation, due both to this cause and cyclic variation, in operating conditions was ±4 percent (see discussion under Precision).

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